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RESEARCH MEMORANDUM

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I - ROTOR DESIGN AND OVER-ALL PERFORMANCE AT TIP SPEEDS FROM 60 TO 100 PERCENT OF DESIGN

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Cleveland, Ohio

NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

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SUMMARY

An inlet-type axial-flow compressor rotor having a hub-tip diameter ratio of 0.4 at the rotor leading edge was designed, fabricated, and experimentally investigated. The rotor was designed for an axial inlet Mach number of 0.6, a tip speed of 1000 feet per second, and a total-pressure ratio of 1.35. These values resulted in a design specific weight flow of 34.9 pounds per second per square foot frontal area (with no allowance for inlet boundary-layer blockage) and a maximum rotor relative inlet Mach number of 1.10.

At design corrected tip speed (1000 ft/sec) and the design total-pressure ratio of 1.35, a corrected specific weight flow of 34.3 pounds per second per square foot frontal area was measured with an adiabatic efficiency of 0.88. Peak efficiencies at speeds lower than design were above 0.95. The inlet boundary-layer blockage factor was found to be 0.98. When the ideal design flow was adjusted for blockage, calculated and measured design conditions (pressure ratio and efficiency) were in good agreement.

The results of the investigation indicated that stages having inlet hub-tip ratios as low as 0.4 can be designed within known practical aerodynamic specifications to produce satisfactory values of stage pressure ratio and efficiency at high specific weight flows.

INTRODUCTION

For aircraft propulsion at high flight speeds, turbojet engines having minimum component dimensions must be developed to obtain a low ratio of weight to thrust. Multistage axial-flow compressors for these engines should be designed for the highest practical values of stage

pressure ratio and weight flow per unit frontal area (specific weight flow) because any increases in the values of these parameters will be reflected in reduced compressor weight and over-all dimensions. There are two methods of increasing the specific weight flow - either the compressor inlet axial velocity can be increased or the hub-tip diameter ratio can be decreased. In recent experimental investigations of inlet-type axial-flow compressor stages with hub-tip diameter ratios of approximately 0.50, considerable improvements in stage pressure ratio and weight flow per unit frontal area were obtained without sacrificing stage efficiency by increasing the design values of axial inlet Mach number and by increasing the allowable values of rotor relative inlet Mach number (refs. 1 and 2). 3087

The primary purpose of the present investigation was to determine the effect of reducing the hub-tip ratio to a value of 0.4 on the performance of a rotor designed with axial Mach number, relative inlet Mach number, and blade loading similar to those reported in reference 1. According to references 1 and 2, no adverse effects on efficiency should be expected.

The degree of design control which could be maintained was also investigated. A compressor with a tip diameter of 14 inches was designed to produce a pressure ratio of 1.35 at a corrected tip speed of 1000 feet per second with no inlet guide vanes and an axial inlet Mach number of 0.6. These values gave a corrected specific weight flow of 34.9 pounds per second per square foot frontal area without correction for boundary-layer blockage in the inlet.

The rotor of this stage was installed and tested in a variable-component axial-flow compressor at the NACA Lewis laboratory. The design procedure used and the over-all performance of this rotor are discussed.

ROTOR DESIGN

There were, of course, two phases of the rotor design, the calculation of the velocity diagrams from prescribed specifications (such as pressure ratio and mass flow) and the selection of blading to establish the velocity diagrams. Since the original purpose of this design was to investigate the performance of a low hub-tip ratio rotor, considerable care was taken in both the velocity diagram calculations and blade selection to ensure good performance barring any unknown adverse effects of low hub-tip ratio turbomachines. The velocity diagram components were set at values known to be practicable.

Velocity diagram calculations. - Before the velocity diagram calculation was begun, three limits were prescribed as follows:

1. The design diffusion factor D (completely described in ref. 3 and defined in the appendix herein) at the rotor tip was not to exceed a value of 0.40.

2. The rotor inlet relative tip Mach number was not to exceed a value of 1.10.

3. The rotor exit (stator entrance) absolute Mach number was not to exceed a value of 0.80.

It should be emphasized that these values, particularly those associated with Mach numbers, do not necessarily represent absolute design limits but rather are proven values commensurate with good stage performance, as demonstrated in references 1 and 2. As in reference 1, no guide vanes were utilized in conjunction with this rotor. A corrected tip speed of 1000 feet per second and radially constant axial inlet Mach number of 0.60 were selected. The maximum relative inlet Mach number was therefore 1.10 at the tip and the relative inlet air angle varied from 31.7° at the hub to 57.1° at the tip. These values resulted in a corrected specific weight flow of 34.9 pounds per second per square foot frontal area, with no allowance for blockage due to inlet boundary layer. A constant energy addition at all radii (vortex-type) through the rotor was utilized, and simple radial equilibrium of static pressures was assumed to give a constant axial velocity at the rotor outlet. With a maximum diffusion factor of 0.40 and a maximum absolute exit Mach number of 0.80, the rotor energy addition was determined to produce a total-pressure ratio of 1.35 with an assumed efficiency of 0.90 at all radii. This design was thus considered to provide a reasonable balance between diffusion factor and stator inlet (rotor outlet) Mach number. Computed density and velocity values for the rotor outlet were used to set the hub radius to maintain the axial velocity leaving the rotor equal to that entering the rotor. Then an estimated 3 percent increase in the calculated rotor outlet area was made to correct for wall boundary layers. The rotor outlet hub-tip diameter ratio obtained accordingly was 0.521. Computed velocity diagram values for assumed stream surfaces passing through radii corresponding to 0, 25, 50, 75, and 100 percent of the passage height at the rotor inlet and outlet are shown in figure 1.

Blade design. - The blade profiles used were made up of circular arcs of different radii for the suction and pressure surfaces and were laid out along the assumed stream surfaces. On the basis of experience and references 1 and 2, an incidence angle of 3° was selected for all blade sections at the design inlet Mach number. The blade camber angle required to obtain the desired turning was determined from preliminary deviation angle data

based on the investigations of references 1 and 2. The design deviation angles thus determined are given in figure 2. A straight line variation of maximum blade thickness from 5 percent of the chord length at the tip to 8 percent of the chord length at the hub was used. This thickness distribution, with a 2-inch chord length, and the required blade camber angles were used to calculate the radii of the circular arcs for the blade suction and pressure surfaces. Leading- and trailing-edge radii were 0.010 inch at all sections. A summary of the design values for the blading is given in figure 2, and a photograph of the rotor disk and blading as tested is shown in figure 3. 3087

COMPRESSOR INSTALLATION

A schematic diagram of the compressor installation is shown in figure 4. Atmospheric air was drawn through a thin-plate inlet orifice, an orifice tank, and an air-operated butterfly valve into a 72-inch-diameter inlet tank. A series of screens and layers of filter paper was installed approximately 60 inches upstream of the compressor inlet bellmouth to provide a smooth and uniform flow at the inlet. Air leaving the compressor entered an annular collector and passed through two outlet pipes spaced 180° apart into the laboratory altitude-exhaust system. An electrically operated gate valve in the outlet piping was used with the inlet valve to control the compressor weight flow and inlet pressure.

Power for the compressor was supplied by a 1500 horsepower, 3587 rpm induction motor through a speed-increasing gearbox with a speed ratio of 9.793. Compressor speed was controlled by varying the frequency of the alternating current supplied to the motor.

INSTRUMENTATION

Air weight flow through the compressor was determined with a thin-plate inlet orifice. Orifice air temperature was measured by four iron-constantan thermocouples, and the orifice pressure drop was indicated by a micromanometer in inches of water.

Inlet tank temperature and pressure were measured approximately 21 inches upstream of the compressor inlet bellmouth by five iron-constantan thermocouple probes and five wall static-pressure taps spaced around the tank circumference (station 0, fig. 4). Lengths of the thermocouple probes were varied so that measurements were made at the centers of equal annular areas. Measured values of temperature and pressure were assumed equal to stagnation conditions for calculation of compressor performance.

Rotor inlet conditions were measured at station 1, about 1/2 inch upstream of the rotor leading edge. From detailed radial surveys of flow angle, total pressures, and static pressures, weight flow was calculated to check the orifice measurement. Values agreed within 2 percent.

At the compressor outlet measuring station (station 2), approximately 4 inches downstream from the rotor trailing edge, total-pressure and total-temperature measurements were made at radial stations located at 10,
3087 30, 50, 70, and 90 percent of the passage height from the outer wall. Three temperature probes and three total-pressure probes were spaced around the compressor. Static-pressure taps were located on the outer and inner walls. A picture of the survey instrumentation is shown in figure 5.

All temperatures were read with a calibrated self-balancing potentiometer and pressures were measured in inches of acetylene tetrabromide. Compressor speed was indicated by a chronometric tachometer.

A magnetic-type blade vibration pickup was mounted in the compressor casing near the rotor-blade leading edge. This pickup was used to give a qualitative indication of the presence of blade-tip vibration.

PROCEDURE

Data points were taken at corrected tip speeds of 600, 700, 800, 900, and 1000 feet per second. At each speed the inlet pressure was maintained at 25 inches of mercury absolute, and the weight flow was varied from the maximum obtainable to a value where blade vibration was encountered. This procedure was adopted to assure the obtaining of data over the complete speed range in the vibration-free region. Obtaining data at lower flow rates was deferred to subsequent investigations. Over-all rotor total-pressure ratio was calculated as the ratio of the arithmetically averaged compressor outlet total pressure to the inlet tank pressure. This value was used with arithmetically averaged inlet-tank and compressor-outlet total temperatures to determine the rotor adiabatic efficiency. The equation used was

$$\eta_{ad} = \frac{T_0\left[\left(\frac{P_2}{P_0}\right)^{\frac{\gamma-1}{\gamma}} - 1\right]}{T_2 - T_0}$$

This method of evaluating compressor performance does not completely consider the flow phenomenon near the rotor blade end regions (hub and tip). Therefore, it is expected that the performance rated on the basis of a mass-weighted integrated average pressure ratio and efficiency based on measurements throughout the wall boundary layers as well as in the free stream would be somewhat different.

RESULTS AND DISCUSSION

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Over-all rotor performance. - Figure 6 shows the over-all performance of the 0.4 hub-tip diameter ratio rotor determined from measurements at the compressor outlet.

At the design corrected tip speed of 1000 feet per second the design total-pressure ratio of 1.35 was measured at a corrected specific weight flow of 34.3 pounds per second per square foot frontal area and an adiabatic efficiency of 0.88. The peak efficiency at design speed was 0.92 at a corrected specific weight flow of 31.5 pounds per second per square foot with a total-pressure ratio of 1.40. Maximum corrected specific weight flow at design speed was 35.2 pounds per second per square foot. Peak efficiencies at tip speeds lower than design varied between 0.95 and 0.98.

Performance at each speed was limited to weight flows above those at which blade vibration was detected. Although the magnitudes of the stresses accompanying the vibration were believed to be small, a decision was made not to investigate the performance in the vibration regions until detailed blade-element data were obtained for the rotor. However, the weight flow operating ranges at each speed reported herein compared favorably with those shown in references 1 and 2.

The data from radial surveys at the compressor inlet were utilized to evaluate the effective flow area reduction due to the inlet wall boundary layer. The boundary-layer blockage factor, which is defined as the ratio of the actual compressor weight flow to the ideal annulus weight flow W_a/W_i or the physical annular area divided by the effective area, was found to have a value of 0.98. This calculation indicated that the (ideal) design specific weight flow was effectively reduced 2 percent to a value of 34.2 pounds per second per square foot frontal area. At a weight flow of 34.2 pounds per second per square foot, the design and measured values of pressure ratio and efficiency were in very close agreement.

SUMMARY OF RESULTS

The following performance results were obtained in a 0.4 hub-tip diameter ratio axial-flow compressor rotor:

1. At the design corrected tip speed of 1000 feet per second, the design total-pressure ratio of 1.35 was measured at a corrected specific weight flow of 34.3 pounds per second per square foot frontal area with an adiabatic efficiency of 0.88. The peak efficiency at design speed was 0.92 at a corrected specific weight flow of 31.5 pounds per second per square foot frontal area with a total-pressure ratio of 1.40. The maximum corrected specific weight flow at design speed was 35.2 pounds per second per square foot. Peak efficiencies at top speeds lower than design varied between 0.95 and 0.98.

2. Surveys at the inlet gave the inlet boundary-layer blockage factor to be 0.98. When the ideal design flow was adjusted for blockage, calculated and measured design conditions (pressure ratio and efficiency) were in good agreement.

CONCLUSION

The results of the investigation indicated that stages having inlet hub-tip ratios as low as 0.4 can be designed within known practical aerodynamic specifications to produce satisfactory values of stage pressure ratio and efficiency at high specific weight flows.

Lewis Flight Propulsion Laboratory
National Advisory Committee for Aeronautics
Cleveland, Ohio, September 9, 1953

APPENDIX - SYMBOLS

The following symbols are used in this report:

A area, sq ft

c blade chord, in.

D diffusion factor, $D = \left[\left(1 - \frac{V_2'}{V_1'}\right) + \frac{V_{\theta,2} - V_{\theta,1}}{2\sigma V_1'}\right]$ 3087

h dimensionless axial velocity, ratio of axial velocity to tip speed, V_x/U_t

i incidence angle, deg

P total pressure, lb/sq ft

r radius, ft

T total temperature, $^{\circ}R$

t maximum blade section thickness, ft

U wheel speed, ft/sec

V velocity, ft/sec

W weight flow, lb/sec

x ratio of tangential velocity to tip speed, V_θ/U_t

y ratio of change in tangential velocity to tip speed

z radius ratio, r/r_t

β air angle (measured from axial direction), deg

γ ratio of specific heats, 1.40

δ ratio of inlet tank total pressure to standard NACA sea-level pressure, $P_0/2116$

Δ deviation angle, deg

η_{ad} adiabatic efficiency

θ ratio of inlet tank total temperature to standard NACA sea-level temperature, $T_0/518.6$

σ solidity, ratio of blade chord to blade spacing

φ camber angle, deg

μ blade setting angle, deg

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Subscripts:

0 inlet tank

1 rotor inlet

2 rotor outlet

CB-2

a actual

f frontal

i ideal

m mean

p pressure surface

s suction surface

t tip

x axial direction

θ tangential direction

Superscript:

' relative to rotor

REFERENCES

1. Lieblein, Seymour, Lewis, George W., Jr., and Sandercock, Donald M.: Experimental Investigation of an Axial-Flow Compressor Inlet Stage Operating at Transonic Relative Inlet Mach Numbers. I - Over-All Performance of Stage with Transonic Rotor and Subsonic Stators up to Rotor Relative Inlet Mach Number of 1.1. NACA RM E52A24, 1952.

2. Robbins, William H., and Glaser, Frederick W.: Investigation of an Axial-Flow-Compressor Rotor with Circular-Arc Blades Operating up to a Rotor-Inlet Relative Mach Number of 1.22. NACA RM E53D24, 1953. 3087

3. Lieblein, Seymour, Schwenk, Francis C., and Broderick, Robert L.: Diffusion Factor for Estimating Losses and Limiting Blade Loadings in Axial-Flow Compressor Blade Elements. NACA RM E53D01, 1953.

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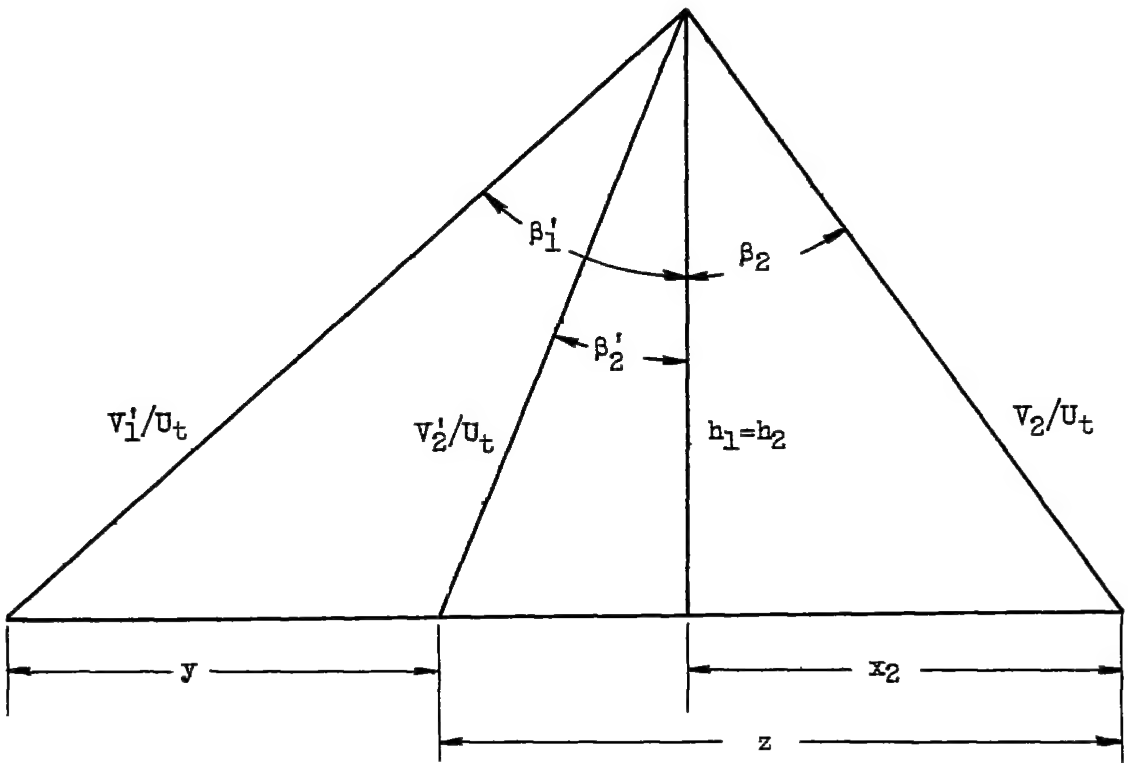
CB-2 back



Direction of rotation, U_t, 1000 ft/sec

Inlet conditions

Percent passage height	z_1	z_2	$h_1=h_2$	$y=x_2$	V_1'/U_t	V_2'/U_t	V_2/U_t	β_1'	β_2'	β_2	D
Tip 100	1.00	1.00	0.647	0.311	1.191	0.945	0.718	57.1	46.8	25.7	0.350
75	.85	.88	.647	.353	1.068	.834	.736	52.7	39.4	28.6	.377
Mean 50	.70	.77	.647	.404	.953	.743	.762	47.3	28.9	31.9	.391
25	.55	.65	.647	.479	.843	.671	.806	40.4	15.1	36.4	.390
Hub 0	.40	.53	.647	.590	.760	.650	.878	31.7	-4.2	42.0	.340

Figure 1. - Design rotor velocity diagrams computed for standard inlet conditions.

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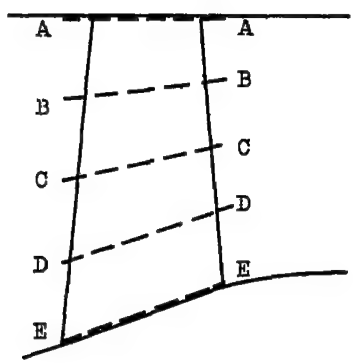
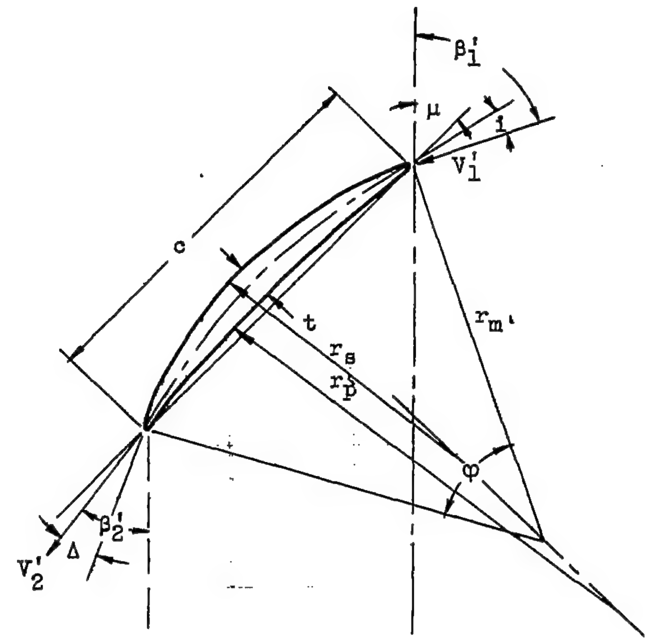


Assumed stream surfaces



Blade geometry

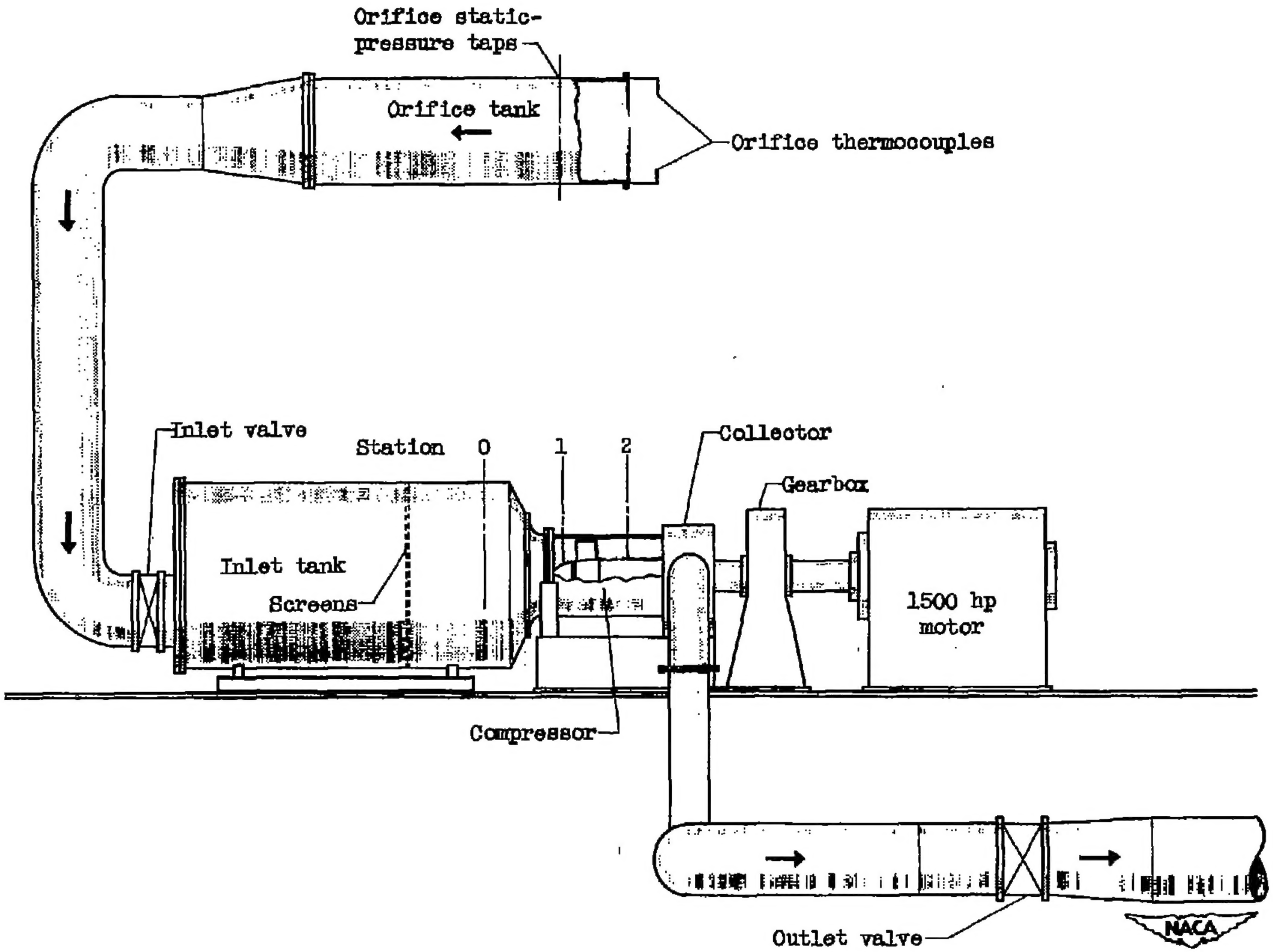
Design values

Percent passage height	Stream surface	z_1	z_2	β_1'	β_2'	Δ	φ	r_m	t/c	σ
100	A-A	1.00	1.000	57.1	46.8	2.1	9.4	12.204	0.0500	0.910
75	B-B	.85	.884	52.7	39.4	4.4	14.7	7.817	.0575	1.050
50	C-C	.70	.767	47.3	28.9	7.1	22.5	5.126	.0650	1.243
25	D-D	.55	.651	40.4	15.1	8.4	30.7	3.778	.0725	1.527
0	E-E	.40	.534	31.7	-4.2	9.2	42.1	2.784	.0800	1.990

Incidence angle, 3.0°
Number of blades, 20
Blade chord, 2.0 in.

Figure 2. - Rotor blade design values and geometry.

Figure 3. - 0.4 hub-tip diameter ratio rotor.



(a) Test rig.

Figure 4. - Schematic diagram of compressor installation.

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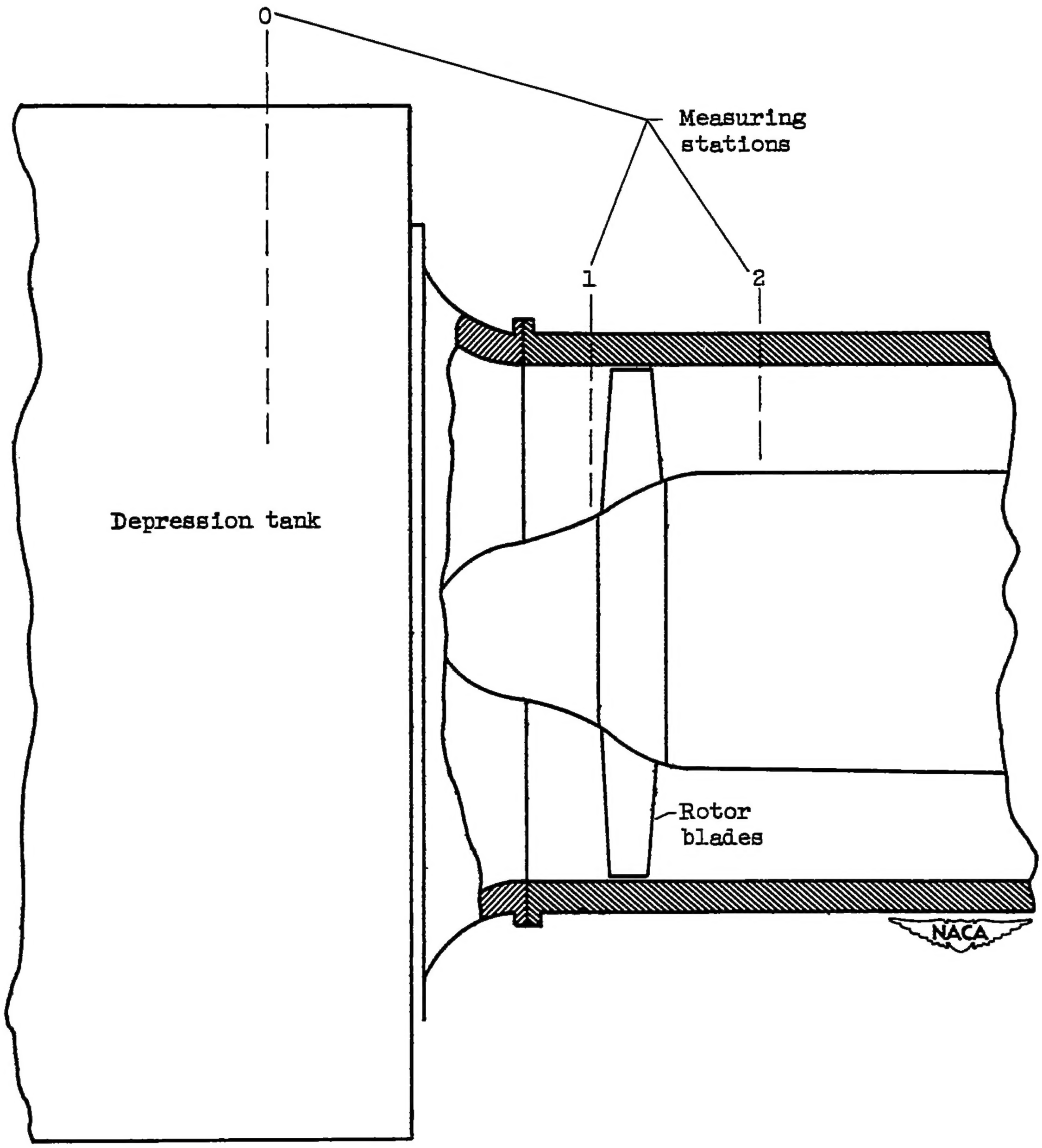
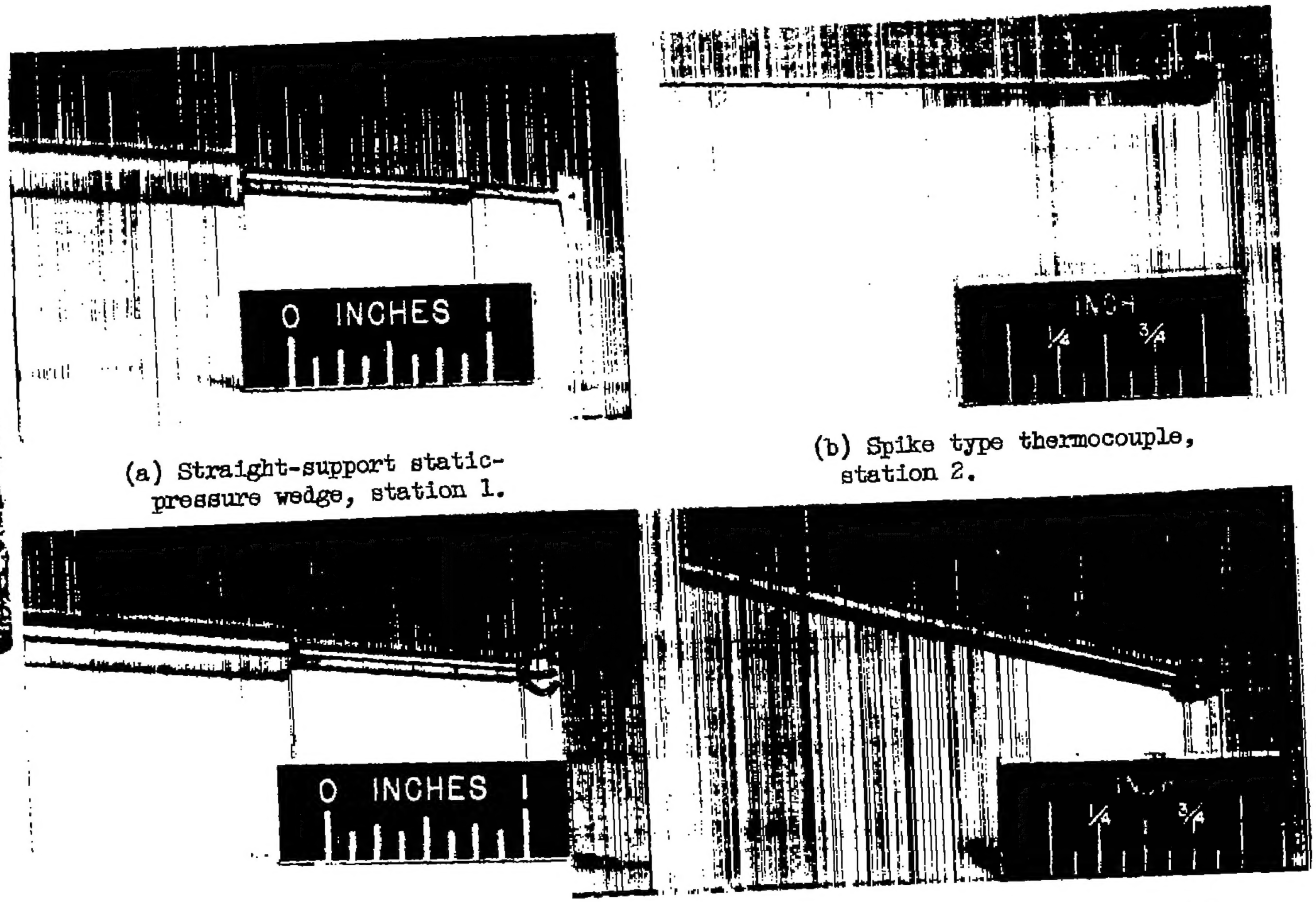


(b) Enlarged view of rotor passage.

Figure 4. - Concluded. Schematic diagram of compressor installation.



(a) Straight-support static-pressure wedge, station 1.

(b) Spike type thermocouple, station 2.

(c) Claw total-pressure tube, station 1.

(d) Shielded total-pressure tube, station 2.

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Figure 5. - Survey instrumentation.

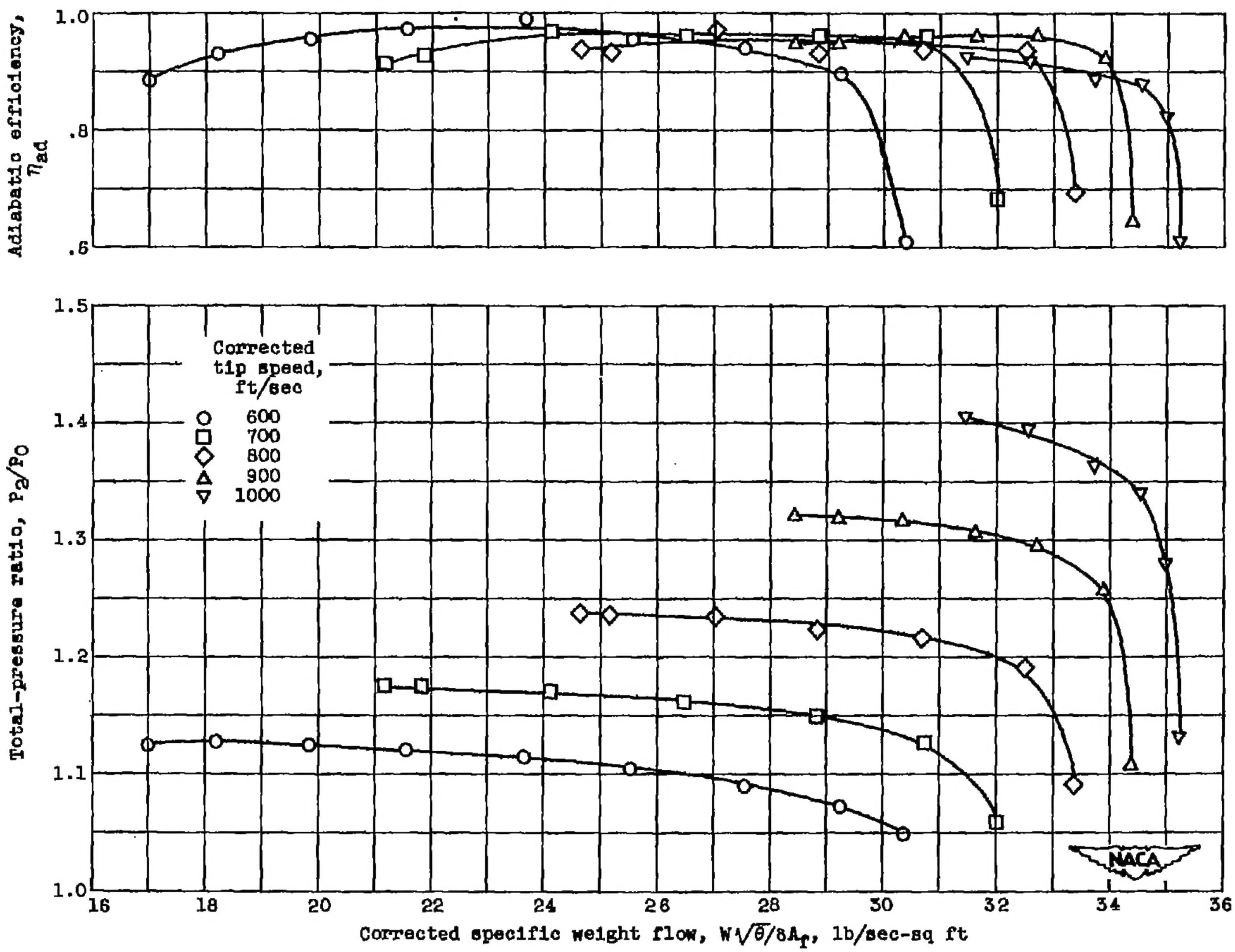

Figure 6. - Over-all performance of 0.4 hub-tip diameter ratio rotor.